Attorney Docket No: 202-126

Hydraulic Radial Bearing

Field of the Invention

The invention relates to a hydraulic radial bearing, a so-called hydro bushing as they are used for mounting engines in motor vehicles.

Background of the Invention

In addition to the insulation of vibrations, which is also provided by conventional rubber elements, the hydro bushings include a damping characteristic for damping the vibrations occurring in the vehicle between the engine and the chassis. As described with respect to FIG. 7, the damping characteristics are achieved with a system integrated into the hydro bushing. This system comprises a support spring, which acts as a piston, and a channel. Here, the mass in the channel and the volume stiffness form a vibration-capable system.

Such systems are matched to approximately 10 Hz and are therefore able to compensate for the inherent vibrations of the engine. Conventional standard hydro bushings are completely unsuited for damping in the lower hearing range.

The invention is directed to a hydro bushing which can filter out acoustic disturbing noises, especially in the region of approximately 130 Hz.

United Kingdom patent application 2,192,968 (corresponding to United States patent application 888,595, filed July 23, 1986) is directed to comparatively large vibration amplitudes in the region of the inherent frequency of the damping system as well as to high frequency vibrations of comparatively small amplitude. For attenuating low frequencies of large amplitude, there are two volume-changeable chambers, which are connected to each other via

a transfer channel as in a standard hydro bushing. Additionally, a further gas chamber for taking up high frequency vibrations of low amplitude is provided and this gas chamber is closed off with an elastic membrane, that is, here, in the acoustic range, only small amplitudes can be filtered out. A further disadvantage is that additional measures are required for damping the expanded range. The manufacture with respect to these measures is associated with additional complexity.

Summary of the Invention

It is an object of the invention to provide a simply configured radial bearing of the kind referred to initially herein which can absorb disturbing noises, especially in the region of approximately 130 Hz, with a low dynamic stiffness.

The hydro bushing of the invention is for radially supporting a motor. The hydro bushing includes: a sleeve-shaped outer body; an inner support body spaced radially from the outer body; a spring body having two legs and being disposed between the outer body and the support body; a volume-changeable work chamber disposed between the legs of the spring body; the volume-changeable work chamber being delimited to the outside by the sleeve-shaped outer body; at least one compensating chamber disposed laterally of the work chamber and having an elastic wall; a transfer channel interconnecting the work chamber and the compensating chamber; the chambers and the channel being filled with a low-viscous hydraulic fluid; the work chamber having an effective cross-sectional area (A_1) and the spring body having a dynamic swell stiffness; the transfer channel having a length (L) and a cross-sectional area (A_2); and, the cross-sectional (A_1), the dynamic swell stiffness, the length (L) and the cross-sectional area (A_2) all being so selected that the hydro

bushing has a natural or resonant frequency of approximately 130 Hz.

The advantages of the invention will be explained hereinafter with respect to a comparison to the relevant state of the art.

<u>Brief Description of the Drawings</u>

The invention will be described in the following with respect to an embodiment of the radial bearing of the invention and a conventional standard hydro bushing wherein:

FIG. 1 is a hydro bushing (radial bearing) according to the invention in a perspective view with this view being from the side and viewed from below at an angle (the outer body is removed);

FIG. 2 is a front view of the radial bearing of FIG. 1;

FIG. 3 is a section view taken along line A-A of FIG. 2;

FIG. 4 is a section view taken along line B-B of FIG. 2;

FIG. 5 is a section view taken along line C-C of FIG. 2;

FIG. 6 is a section view taken along line D-D of FIG. 3;

FIG. 7 shows a standard hydro bushing according to the state of the art viewed from the side at an angle from below (outer body removed); and,

FIGS. 8a and 8b are schematics for explaining the continuity equation and the Bernoulli equation.

<u>Description of the Preferred Embodiments of the Invention</u>

The standard hydro bushing 102 shown in FIG. 7 essentially comprises: a sleeve-shaped outer body 104 (shown in phantom outline); an inner mounting body 106 (for accommodating a bearing lug) which is spaced radially to the outer body 104; and, a two-legged spring body 108 disposed between the outer body 104 and the inner body 106.

A volume-changeable work chamber 110 is filled with hydraulic liquid and is disposed between the legs (108a, 108b) of the spring body 108. The work chamber 110 is delimited from the outside by the outer body 104 and on both sides by respective massive legs (112a, 112b). A transfer channel 114 is arranged annularly at the inner side of the outer body 104 and extends peripherally. One end of the transfer channel 114 has an opening 116 to the work chamber 110 and the other end has an opening 118 to a compensating chamber 120a arranged to one side in the bearing 102. The compensating chamber 120a is likewise delimited toward the outside by the cylindrical body 104 which encloses all. The compensating chamber 120a includes a flexible membrane 122a toward the inside.

For reasons of symmetry, a further compensating chamber 120b and a flexible membrane 122b (not shown) are disposed on the side lying opposite the compensating chamber 120a and are configured overall to be the mirror image thereof. Both compensating chambers (120a, 120b) are connected to each other via a connecting channel 124.

If a dynamic load F_1 acts on the hydro bearing 102, then the two-legged spring body 108 deforms whereby the spring body 108 presses like a piston on the hydraulic liquid disposed in the work chamber 110. The effective piston area A_1 is given by the liquid volume ΔV, which is displaced from or into the "piston", and its speed v_1. The liquid quantity displaced by the spring body 108 is compelled to escape through the transfer channel 114 (cross section a_2, flow speed v_2) into the compensating chambers (120a, 120b). The flow takes place in accordance with the continuity equation:

$$A_1 \cdot v_1 \cdot \rho = a_2 \cdot v_2 \cdot \rho$$

or $A_1 \cdot v_1 \cdot \rho = A_2 \cdot v_2 \cdot \rho$ (see FIG. 8a)

and the Bernoulli equation

$$p_1 + \rho/2 \cdot v_1^2 = p_2 + \rho/2 \cdot v_2^2$$ (see FIG. 8b).

A jump in velocity of the speed v_1 to the speed v_2 takes place at the crossover location from the wide work chamber 110 to the narrow crossover channel 114. This jump in velocity releases considerable reaction forces F_2 (FIG. 8b) which effect a swelling of the spring body 108. The spring body 108 is therefore characterized by a so-called dynamic swell stiffness in addition to static spring stiffness. The dynamic swell stiffness in combination with the effective mass of the hydraulic liquid vibrating in the transfer channel 114 essentially determines the inherent frequency of the hydro bearing 102 which is effective to reduce vibration. This inherent frequency lies at approximately 10 Hz in a conventional standard hydro bearing 102.

With a conventional hydro bearing 102, it is not possible to realize the frequency region of approximately 130 Hz with purely constructive measures (dimensioning measures).

The significant similarities and the differences with respect to the hydro bearing 2 of the invention will now be explained based on a comparison.

The radial bearing 2 of the invention is shown in FIGS. 1 to 6 and essentially likewise includes: a sleeve-shaped outer body 4 (shown in phantom outline in FIG. 1); an inner mounting body 6 which is disposed radially spaced to the outer body 4; and, a two-legged spring body 8 disposed between the outer body 4 and the inner body 6. A work chamber 10 is likewise filled with hydraulic liquid and is changeable in volume. The work chamber 10 is likewise disposed between the legs (8a, 8b) of the spring body and is delimited toward the outside by the outer

body 4.

The bearing 2 of the invention also includes compensating chambers (20a, 20b) which are likewise connected to each other by a connecting channel 24. Each of the compensating chambers (20a, 20b) is provided with an elastic wall (flexible membrane 22a or 22b).

According to the invention, the work chamber 10 is connected to the two compensating chambers (20a, 20b) via respective transfer channels (14a, 14b). These transfer channels (14a, 14b) comprise a partial non-presence of the side walls (see legs 112a and 112b; FIG. 7) of the work chamber 10. The width B of the channels (14a, 14b) is identical to the total height H of the cylindrical bearing 2. The length L of the two channels (14a, 14b) is considerably less than their width B. The channels (14a, 14b) extend directly into the corresponding compensating chambers (20a, 20b) which is favorable with respect to flow. The two transfer channels (14a, 14b) are connected parallel to each other. In this way, their respective cross sections add to a total cross section A_2. The two compensating chambers (20a, 20b) are connected directly via respective transfer channels (14a, 14b) to the volume-changeable work chamber 10. For this reason, the connecting channel 24 functions only to compensate for an asymmetric loading of the bearing 2. The connecting channel 24 bridges the two compensating chambers (20a, 20b).

With the construction in accordance with the invention, a dimensioning of transfer channels is achieved for the first time which makes it possible to place the frequency, which is relevant for the absorption, in the region of approximately 130 Hz. The relevant frequency is here also computed from the effective mass

of the hydraulic liquid, which is vibration capable in the transfer channels, in combination with the dynamic swell stiffness of the spring body (the dynamic swell stiffness is given by the piston cross section A_1 and the flow speed v_1 present in the work chamber).

To further reduce the faulty adaptation between the piston cross section A_1 and the sum of the transfer channel cross-sectional area A_2, the work chamber 10 includes constrictions (26a, 26b).

In order to hold the dynamic stiffness as low as possible, the hydraulic liquid should have a viscosity as low as possible. The viscosity of the liquid and the geometry of the transfer channels (10 <-> 14 <-> 20) should be so selected that the flow of the liquid, which moves from one chamber into the other, is as laminar as possible. This is the case when the Reynold's number is:

$$R_e = \rho \cdot r \cdot v/\eta < 1200$$

wherein: ρ = density of the liquid; η = viscosity; r = characteristic length; v = speed of the liquid.

When realizing the bearing, a viscosity in the range of $\eta = 0.01\ g \cdot cm^{-1} \cdot s^{-1}$ (water, 20° C) to $\eta = 14.9\ g \cdot cm^{-1} \cdot s^{-1}$ (glycerine, 20° C) has been shown to be especially suitable.

Because of practical considerations, a frost protection agent should be added to the water when used as a hydraulic liquid, for example, glycol or glycerine, that is, dihydric alcohol or trihydric alcohol.

Furthermore, it should be noted that there is always an adequate distance to the boiling point of the hydraulic liquid in order to reliably preclude cavitation.

It is understood that the foregoing description is that of

the preferred embodiments of the invention and that various changes and modifications may be made thereto without departing from the spirit and scope of the invention as defined in the appended claims.